Volume 44 SEPTEMBER, 1958 Number 9

Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

This Issue

FREE PISTON ENGINES

PUBLISHED BY
THE TEXAS COMPANY
TEXACO PETROLEUM PRODUCTS

LUBRICATION

A TECHNICAL PUBLICATION DEVOTED TO THE SELECTION AND USE OF LUBRICANTS

Published by

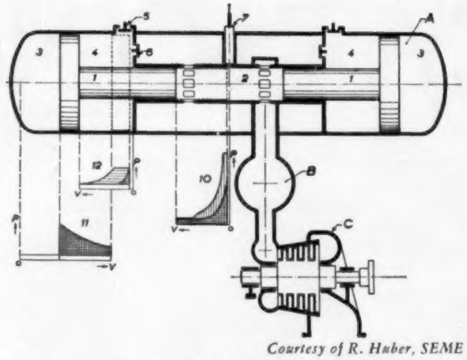
The Texas Company, 135 East 42nd Street, New York 17, N. Y.

Vol. XLIV — September, 1958 — No. 9

FREE PISTON ENGINES

FREE piston engines have been used for a number of years as air compressors in single and multi-stage form in both France and Germany. However, it is only within the last decade that free piston engines have been developed as gasifiers (or gas generators) and combined with gas turbines to generate electrical power, to propel ships and locomotives, and to drive centrifugal pumps and compressors. Considerable effort has also gone into the development of experimental free piston-gas turbine units for use in automobiles, trucks and tractors. The recent accelerated rate in the development of free piston-gas turbine systems and their application to a wide range of power needs has created considerable interest in this relatively new type of power and the extent that it may replace or supplement existing power sources. Also students of engineering colleges and universities have expressed a broad interest in free piston—gas turbine systems in studying the basic principles of internal combustion machines and in the application of creative thinking. The purpose of this issue is to give a brief history of free piston engine developments, to discuss some of their operational features including fuel and lubrication requirements, and to look at possible trends in future developments and applications.

One of the first questions the uninitiated might ask is just what is a free piston engine? Basically the free piston engine is a two stroke, uniflow scavenged, opposed piston diesel engine that is supercharged by direct connected reciprocating compressor pistons as illustrated in Figure 1. In this system all of the reciprocating work produced in the diesel cylinder is absorbed by the compressor pistons and the friction of the moving parts. In addition to supplying air, the compressor pistons also store up energy in "bounce chambers" to stop piston motion at the end of the outward power stroke and to return the pistons to the center position during the inward compression stroke. The opposed piston configuration presents a symmetrically balanced machine where the relative motion of the pistons is largely controlled by pneumatic forces and no crankshaft or flywheel is used. When the free piston engine is used as an air compressor, part of the air from the compressor cylinders goes to supercharge or scavenge the power cylinders at relatively low pressures (2-5 psig) and the remaining air is used for operating air tools or for other pneumatic purposes. When the engine is operated as a "gasifier," all of the air from the compressor cylinders, at relatively high supercharge pressures (40-90 psig), normally passes through the diesel power cylinder where fuel is burned. The resulting high pressure, high temperature exhaust gas may then either be used for some industrial purpose or expanded through a power turbine to produce useful shaft power. At the present time the major interest is in prime mover applications of free piston gasifiers combined with power turbines. However, free piston air compressors still appear attractive and some industrial applications for the high pressure, high temperature exhaust gases may be developed.



Courtesy of R. Huber, SEME

Figure 1 — Diagramatic sketch of a free-piston gas generator and gas turbine showing pressure-volume diagrams of the engine, compressor and cushion cylinders.

WORKING CYCLE OF FREE PISTON ENGINES

In the design of free piston engines, several different arrangements of the component parts and various operating cycles may be considered. The compressor cylinders may be arranged to deliver air on (1) the inward stroke, (2) the outward stroke, or (3) both inward and outward strokes. In any of these cases, provision must be made for a bounce cushion or chamber that will store up energy to stop the pistons on the outward stroke and return the pistons toward the center of the cylinder. On inward compressing machines the outer side of the compressor piston may act as the bounce cushion. In outward compressing engines the air cushion is normally formed by separate stepped down bounce pistons. Here the inner side of the compressor cylinder may also be used as a reverse bounce chamber. Stepped down bounce pistons can also be used with double acting compressors. In two stage double acting compressor design, the first stage of compression is accomplished on the inward stroke and the second stage on the outward stroke with the bounce chamber formed on the outer side of the compressor piston. Compressor cylinders may also be arranged for additional stages of compression such as the four stage Junkers air compressor.[6]

Gasifier components may also be designed for different operating cycles:[24]

1. Simple cycle using single stage compression.
2. Simple cycle with aftercooling of the scavenging air.
3. Two stage reciprocating compression with intercooling.
4. Two stage reciprocating compression with intercooling and aftercooling.
5. Turbocharged, intercooled, one-stage reciprocating compression.
6. Turbocharged, intercooled, one-stage reciprocating compression and aftercooled.
7. The foregoing cycles with reheat.
8. The foregoing cycles with wet compression.

Of these various cycles and component arrangements, the simple cycle, single stage, inward compressing type is used in the Pescara system which is representative of the majority of commercial free piston gasifiers in service at the present time. The arrangement of this type free piston unit may be described by referring to Figure 1. The gas generator A contains two opposed piston assemblies (1) with the diesel power cylinder (2) in the center. The two single acting compressor cylinders (4) are located on each end of the central housing. The end spaces (3) constitute the cushion or "bounce" cylinders which store the energy for the return stroke. Fresh air is drawn through the suction valves (5) and is delivered through the discharge valves (6) into the air box which surrounds the engine cylinder. The fuel is injected by several injectors (7) mounted in the central plane of the combustion chamber.

Figure 1 may also be used to describe the processes that occur during each cycle of the piston motion. In order to start the engine, the pistons are first positioned at the outer dead point (ODP) as illustrated. Highly compressed air from a starting air supply is introduced into the bounce chambers (3) which drives the pistons towards the center, compressing air in both the compressor cylinders and the diesel power cylinder. Fuel is injected as the pistons approach their inner dead position (IDP) and combustion takes place. As the compressor pistons approach their IDP, the discharge valves (6) open to admit air to the engine case or air box. The pressure of the gases in the combustion chamber of the power cylinder added to the pressure of the air remaining in the clearance space of the compressor cylinders drives the pistons outwards, compressing air in the cushion cylinders (3). The work done during the inward stroke is represented by the vertically hatched areas in the pressure volume diagrams (10) and (12) for the engine and compressor cylinders. This work is subsequently recovered, except for friction losses, as shown by the pressure volume diagram (11) for the cushion cylinder. As the power pistons move outward, the exhaust ports, as shown on the right, are uncovered first followed by uncovering the scavenging ports located on the left. This allows air from the engine case to scavenge the combustion gases from the power cylinder. The resultant exhaust gases combined with the excess scavenging air then flows through a gas collector (B) which serves to smooth out the pulsations to the power turbine (C) where the gases are expanded to atmospheric pressure. As soon as the pressure in the compressor cylinders (4) has dropped to about atmospheric pressure the automatic suction

[6] For this and subsequent references see corresponding number in bibliography at end of article.

valves (5) open and fresh air enters the cylinders. As the pistons approach the outer dead point, the buildup of pressure in the cushion chambers stops outer piston travel and the inward compression stroke begins. When the power pistons cover the exhaust and intake ports the scavenging process ends. At this point the trapped air in the power cylinder is at the same pressure as the gas collector since the exhaust ports close last on the inward stroke. As compression continues the pistons come to rest at the inner dead point and the cycle repeats. The energy stored in the cushion cylinders is expended in providing the work done in the compression cylinders and the work of compression in the power cylinder as well as the work associated with the friction losses. The work of the inward stroke is illustrated by the horizontally cross-hatched areas on the pressure-volume diagrams.

In one respect the free piston-gas turbine power plant may be considered as one extreme of a highly supercharged compound engine which combines the high thermal efficiency of the diesel engine combustion cycle with the simplicity, flexibility and good torque-speed characteristics of the relatively light weight and compact power take-off turbine. This concept has been followed to some degree in the evolutionary development of the highly efficient turbo-charged diesel and gasoline engines where part of the energy in the exhaust gas is used to drive a gas turbine connected to a centrifugal air compressor. This in turn has presented the evolutionary idea of connecting the turbo-charger to the engine crankshaft to recover shaft power from the exhaust gas. For example, in the Turbo-Compound aviation engine, from 15 to 20% of the propeller shaft power is obtained from the exhaust driven turbines which results in 10% lower specific weight and a 20% lower cruise specific fuel consumption. The next step, which consists of developing all of the useful power from the turbine, leads to the gasifier concept. Here the gasifier may be a diesel or gasoline engine and the compressor either reciprocating or rotary. The selection of a free piston engine as the gasifier simplifies construction as the crankshaft, connecting rods and related bearings can be eliminated. This arrangement also permits the stroke and the dead center positions of the pistons to be adjusted to the operating conditions of the unit so that the pressures in the power cylinder may be kept within permissible limits over a wide range of supercharge air flow rates and gas delivery pressures. Here the maximum power developed in the diesel cylinder at high supercharging pressures is not limited by permissible bearing loads as may be the case in some conventional diesel engines but eventually becomes more a function of permissible thermal loading and ring pressures.

The gasifier operates at high compression ratios up to 50 to 1 and therefore has a high thermal efficiency in the range of 40-45% at full load. With a turbine efficiency of 80% this would give an overall efficiency of 32-36% at the turbine shaft which approaches that of a conventional diesel engine. At these compression ratios and high excess air conditions, the free piston gasifier can successfully burn a wide variety of petroleum fuels ranging from high octane gasoline to heavy residual Bunker C type fuel in addition to whale oil, cotton seed oil and peanut oil.[1] Here power output varies directly with the heat content of the fuel used. Another important feature of the gasifier system is that the temperature of the exhaust gases entering the power turbine is relatively low (800-1000°F.) compared with 1250-1500°F. in conventional gas turbines. This is due to the fact that part of the energy produced during combustion in the power cylinder is absorbed by the compression of supercharge air. The combustion gases then are first cooled during initial expansion in the power cylinder which is followed by additional cooling from mixing with the supercharge air during the scavenging period. These lower exhaust gas temperatures permit the use of less expensive non-critical materials in the construction of the power turbine and reduce thermal distortion problems. By the same token the lower inlet temperatures minimize deposit buildup and corrosion of turbine blades and nozzles from vanadium and other ash components when residual fuels are burned. At the same time, the temperature in the gasifier is maintained sufficiently high to minimize corrosion from the sulfur that may be present in the fuel. Since the gasifier is not mechanically coupled to the power turbine, the units may be arranged in the most convenient way for each application. This flexible arrangement also permits several gasifiers to be connected to one turbine to obtain the desired power output. When more than one gasifier is connected to a turbine it then becomes possible to shut one unit down for repairs while the remaining gasifiers carry part or all of the load. Maintenance is simplified by the reduction of moving parts. Since the moving parts of the free piston gasifier are in near perfect balance, vibration loads are practically negligible. This simplifies foundation requirements. Overall power to weight ratio of the free piston-gas turbine plant is comparable to or better than other conventional prime movers and the overall space requirement is generally less. These are some of the features of free piston machinery. More detailed information on these and other operational characteristics will be given in the discussion to follow.

HISTORY

The principle of the free piston engine has been known for some time and the serious historian may find numerous references to early free piston appli-

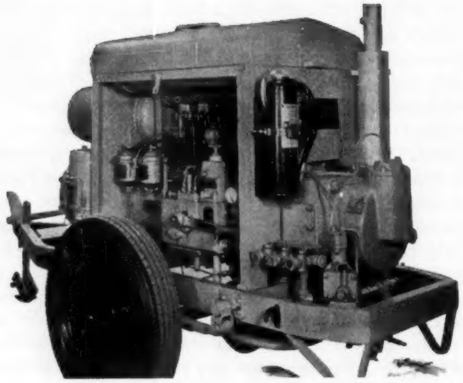


Figure 2 — Sigma P-13 type 60 HP free-piston air compressor.

cations in the literature. For example, in 1857 Barsanti and Matteucci proposed a free piston gas engine[2] which consisted of a vertical cylinder, open at the top and containing a heavy piston that was coupled by means of a chain, to a rachet wheel. The explosion of a gas/air mixture beneath the piston drove it rapidly upward, so that a partial vacuum was formed in the cylinder. The vacuum combined with the weight of the piston, did mechanical work on the down stroke of the piston. Thus the motion of the piston was not constrained, at least during the combustion stroke. While this proposal failed to develop into a practical engine, some ten years later Otto and Langlen exhibited a very similar engine at the 1867 Paris Exposition. Under their persistent efforts this engine for the next ten years proved to be the most economical and reliable but noisiest gas engine on the market until it was superseded by the Otto "silent type" trunk piston gas engine. In addition to these early efforts the steam boiler feed water pump and the Humphrey pump may also have inspired inventors to apply the internal combustion engine to the free piston principle. Professor Junkers is reported to have given a talk before a German naval engineering society in 1912 in which he described the use of a free piston apparatus to do research on heat transfer relationships in internal combustion engines[3]. In the intervening years, British, French and German patents[4] on free piston applications were filed, however, references to related development work are obscure.

Shortly after World War I, Marquis Raul de Pateras Pescara was searching for a light weight engine compressor unit to provide compressed air for propelling a helicopter rotor by means of reaction jets[6, 37]. Since there were no existing machines of this type available, he developed his own design in the form of a free piston engine compressor unit. While this machine did not develop into a practical helicopter application, it later evolved into a commercial free piston compressor unit and led to the free piston gasifier concept where as previously noted *all* of the air from the compressor is used to supercharge the diesel cylinder and the resulting exhaust gases are expanded through a power turbine to provide useful work.

Pescara started development work on free piston engines around 1922[4] and shortly thereafter he assembled a group of engineers under the direction of R. Huber to design free piston machinery. The first experimental compressor was built in Paris about 1925. Around the same time it is understood that the first free piston compressors actually built and used were made by Junkers in Germany and Breguet in France. During this early period, development was primarily limited to free piston air compressors and several types were constructed before 1939. In 1941 the Société Industrielle Générale de Mécanique Appliqué, (hereafter designated "SIGMA,") a sub-licensee of the Pescara patents, started production of the 60 h.p. type P-13 compressor of which some 1200 units are in commercial use throughout the world. In fact one of these SIGMA P-13 units as illustrated in Figure 2 has been installed in one of a major petroleum refiner's laboratories to study its operational characteristics and to develop information on fuel and lubricant requirements.

Design work on free piston gasifiers began around 1933 and in 1937 the Societe d'Etudes et de Participations, "SEP," of Geneva, Switzerland became one of the major licensees of the Pescara patents, under Auto Compresseurs Pescara ("A.C.P."), that included exclusive manufacturing rights for the world except for the British Commonwealth, Eire, and Egypt[4]. Licensee arrangements for these latter areas were assigned by A.C.P. to Alan Muntz and Co., Ltd. of Middlesex, England. In 1938 the Societe d'Etudes Mecaniques et Energetiques, "SEME" was formed by SEP as a design and development group under the direction of R. Huber. "SIGMA" of Paris, France, now under the direction of D. Coste, became the leading sub-licensee under SEP in 1937. The SIGMA group, in association with SEME, have largely been responsible for the development of the GS-34 gasifier, nominally rated at 1250 gas h.p.

Beginning in 1937 several different types of gasifiers were investigated by the SEME-SIGMA group but the first machine, rated at 850 gas h.p., was built and tested at the Belfort works of the French company, Alsthom, in 1939[38]. Two of these gasifiers were connected to one 950 kw turbo generator set. Tests on this set were interrupted by the war, however additional types of gasifiers were built and tested later at the SIGMA Venissieux works and the prototype of the present model GS-34 unit was ordered by the French Admiralty and placed in service in July 1946[38]. Since this time over 18 organiza-

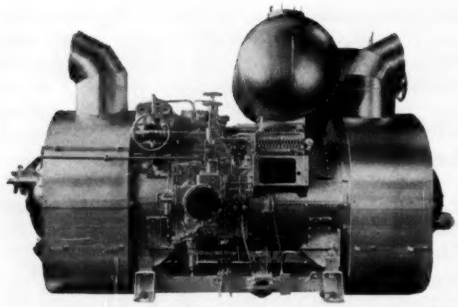
tions have taken out sub-licensee agreements[4] with either SEP or Alan Muntz and Co. Limited. These firms include seven French, two Dutch, one German, seven United Kingdom and one American. Pescara has been credited for his pioneering efforts that have resulted in commercial application of the free piston engine[6]. Recognition has also been given to Huber and his associates for their persistent efforts in carrying out engineering design studies in addition to Professor G. Eickelberg who was a valued consultant during the development period and D. Coste who was a strong proponent for applying the free-piston-gas turbine as a marine prime mover[6, 26]. Financial support from the French Admiralty, and the Merchant Marine and Electricite de France also contributed to the realization of the SIGMA-SEMA GS-34 development[6, 38]. A more detailed description of the GS-34 and its applications will be given in a following section.

While practically all of the commercial free piston engines in service today apparently stem from the original Pescara patents, considerable efforts have been made on the design and development of these machines in other countries. As previously mentioned, Junkers in Germany was one of the early proponents of the free piston engine concept. This led to the development of 5 models of free piston compressors ranging from a single stage machine delivering 425 cfm of free air at 57 psig to a 4 stage compressor delivering 70 cfm of free air at 3000 psig[3, 6, 7, 8, 9]. The 4 stage 3000 psig unit was adopted by the German Admiralty and was extensively used during World War II for providing compressed air for launching torpedos[6]. It is understood the other 4 models including a 100 psig, 120 cfm, 2 stage portable mounted compressor, are currently marketed by the Junkers organization. Shortly after Pearl Harbor, the Japanese manufactured an identical model of the high pressure Junkers compressor for naval use[6, 7].

In Switzerland, Sulzer Brothers started work on free piston engines around 1936 in conjunction with a broad program to determine the most promising line of development for internal combustion engines[18]. This work led to the construction and test of a turbo-charged single stage, outward compressing free piston-gas turbine unit which was rated about 2000 hp at the turbine shaft. As a result of the tests on this single unit, a 6000 hp plant was built which was composed of 3 gasifiers and a single turbine. While performance of this plant was considered very good, Sulzer later abandoned the project in view of the promising future indicated for the combustion gas turbine[18].

In England, the Alan Muntz and Co. became a licensee of the Pescara patents in 1937 to manufacture and sell free piston units in the United Kingdom, Eire and Egypt[4]. Their development work was first concentrated on a free piston compressor now designated as the P-42 which is rated at 100 cfm, 90 psig and 23 hp[10]. This machine is similar to the SIGMA P-13 design. The major difference is that the Muntz design uses a helical spring for starting while compressed air is employed in the P-13 model. Later the Muntz Co. started an independent development on a free piston gasifier now known as the CS-75 model. This unit is a smaller version of the SIGMA GS-34 and is rated at 420 gas hp[11]. The Muntz CS-75 has now completed several thousand hours of development running and two units have been ordered for shipboard auxiliary generating sets. In addition to the CS-75 development the Muntz Company are also exploiting the manufacture and sale of SIGMA GS-34 gasifiers under their licensee agreement with the Pescara ACP group and they have arranged sub-licensee agreements with some 8 different manufacturers[4].

In the United States, it has been reported that Pratt and Whitney Aircraft Company initiated a program to develop a free piston aircraft power plant in 1941. However, the rapid development of the aircraft turbojet, plus the difficulties in adapting a free piston engine for high altitude operation led to the abandonment of this project in 1946[6, 12]. Around 1942 the U. S. Navy initiated a study of the possibilities of free piston engines after acquiring a Junkers compressor. Several performance and design studies were made[6] and a limited number of prototype compressors of the Junkers type were constructed by several engine builders including Worthington Corporation[9], and Clark Brothers. In 1943 Baldwin-Lima-Hamilton Corporation started development work on free piston gas generators suitable for naval propulsion purposes under a contract with the U. S. Navy[13]. The first compressor, designated as Model A, was a high output outward compressing horizontal type to produce 700 gas hp at 105 psia and 1350°F. The second machine, Model B, was similar to the Model A. but produced 802 gas hp at 105 psia and 1342°F. In 1949 a vertical Model D was designed primarily for locomotive use where six units were to be connected to a single gas turbine. This led to the model DL horizontal single stage inward compressing type developed for the U. S. Navy Bureau of Ships[14, 15] rated at 650 gas hp at 62 psig and 1054°F. exhaust pressure and temperature which was later turbo supercharged to 1000 gas hp at 86 psig and 980°F. The most recent Baldwin-Lima-Hamilton development is the Hamilton FP-165[15, 16] which is a two stage compressing type rated at 250 horsepower at the turbine shaft when turbocharged or 150 shaft horsepower without turbocharging. A radial inflow turbine coupled to an automatic speed reducing transmission is proposed for this unit. Either single or multiple gasifier units are proposed to provide from 125 up to 1000 shaft



Courtesy of Cleveland Diesel Engine Division, GMC

Figure 3 — General Motors Model GM-14 free-piston gasifier.

horsepower for use in stationary and mobile power plants, railroad locomotives, marine power plants, farm tractors, and other heavy duty vehicles.

Early in 1942 the Cooper Bessemer Co.[17] began laboratory test investigation of free piston engines which resulted in the design and construction of their Model R horizontal single stage outward compressing unit rated at approximately 1750 gas hp at 72 psig and 1000°F. exhaust gas pressure and temperature. A supercharged version was also considered which would be rated at 3000 gas hp. These units in both single and multi-unit combinations were proposed for electrical power generation, pipe line pumping and marine propulsion. However, this project was later abandoned in view of advances made in their turbocharged diesel and gas engines.

In 1953 General Motors Corporation became actively interested in the possibilities of free piston engines.[1] Through a working agreement with SIGMA and SEME, tests were first carried out on the Sigma P-13 and Muntz P-42 air compressors. Later, a SIGMA GS-34 gasifier was obtained and several thousand hours test experience has been accumulated with this unit, a considerable portion of this using a heavy residual Bunker C type fuel containing about 3.5% sulfur[1]. As a result of these evaluation tests, several modifications were incorporated to improve the output and durability of the GS-34 unit which evolved into the Model GM-14 gasifier as illustrated in Figure 3. In 1954, the Cleveland Diesel Engine Division of General Motors was awarded a contract by the U. S. Maritime Administration for a 6000 hp free piston-gas turbine power plant to be installed in the Liberty ship "William Patterson"[19]. This contract was made in connection with the Administration's Liberty Ship Conversion and Engine Improvement Program which included the conversion of four ships with the following different types of main propulsion machinery in the 6000 shaft hp range: steam turbine, supercharged diesel engine, open cycle gas turbine, and a free piston gas turbine unit. It was proposed that all of these power plants be designed to operate on heavy residual fuel oil.

Essentially, the 6000 hp unit consists of six gasifiers each rated at approximately 1230 gas hp connected with two 3000 shaft hp gas turbines driving a single reduction gear and single propeller. After completion of successful dock trials, the "William Patterson" was placed in service in October 1957. The experience developed with the free piston machinery in this vessel as compared with that of the other three types of power plants that have been installed could form the basis for any future Liberty ship re-powering program. It may also lead to further adoption of new types of power plants for commercially operated vessels. Accordingly the operating experience of these vessels is being followed with considerable interest.

In addition to the Liberty ship application, General Motors Research Division in cooperation with SEME developed and built the GMR-4-4 "Hyprex" free piston gas turbine power plant that was installed in their experimental XP-500 automobile in 1956[20]. The GM Hyprex engine presented a new approach in free piston engine design in which two single stage, inward compressing free piston gasifiers were joined or "twinned" together as a siamesed unit[20].

While complete performance data for the Hyprex engine is not available, the output is based on a nominal 250 gas hp and a specific fuel consumption value under 0.45 lb. per gas horsepower hour was reported during preliminary testing[20]. The nominal dimensions of the gasifier are 40 inches long, 34 inches wide, and 18 inches high. The two power cylinders have a bore of 4 inches with a piston stroke of approximately 5 inches. Compressor cylinder bore is 11 inches. Maximum cyclic speed is 2400 strokes per minute and minimum speed is in the order of 1000 cycles per minute. As the small free piston engine power plant is still in the experimental stage, further development and evaluation is indicated for possible automotive applications. A rather extensive study has also been carried out by General Motors Research to determine the more desirable basic engine dimensions and to predict operating characteristics of a given engine[23]. This work included a thermodynamic study of piston motion by the simulation of a free piston engine with a digital computer which should be of value in any future designs.

The Ford Motor Company started active development work on free piston engines in 1954[21]. The objective of the initial program was to obtain basic thermodynamic and design information on small high speed free piston engines which was to be a basis for designing industrial or automotive size power plants. The first Ford free piston gasifier had a 2 inch diameter power cylinder, and a 5 inch compressor cylinder which developed up to 16 gas hp at 3600 cycles per minute. This unit was a single

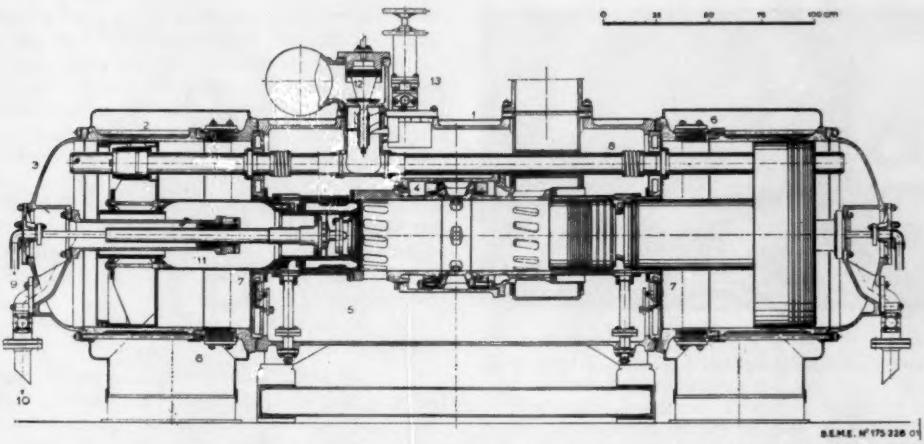


Figure 4 — Vertical cross section of a SIGMA Model GS-34 gasifier.

stage inward compressing type and ran on various liquid fuels ranging from 0 to 100 octane number in addition to gaseous propane. Operation was reported successful with either carburetion or fuel injection, and with either spark ignition or compression ignition. Based on information gained from this gasifier, a larger version rated at approximately 150 gas hp was constructed and designated as Model 519. The basic geometry for this unit was largely determined from performance analysis programmed on the IBM 702 computer. The Model 519 gasifier is a single stage inward compressing machine with a power cylinder bore of 3.75 inches, compressor bore of 11", minimum effective stroke 4.2 inches and a maximum stroke 6.8 inches[22]. The Model 519 is reported capable of developing about 150 gas hp at a discharge pressure ratio of 3.6 and a cyclic speed of 2400 cycles per minutes with a specific fuel consumption of around 0.45 lbs./gas hp-hour. The gasifier is approximately 37 inches long and 16 inches in diameter. Following evaluation studies, the 519 gasifier, coupled to a 5½ inch diameter, 43,000 rpm, radial inflow turbine and a reduction gear was used in the construction of the Ford Typhoon Tractor[21].

In 1955 the Batelle Memorial Institute began studies on a natural gas fueled free piston engine compressor for air conditioning purposes under a program jointly sponsored by the Remington Arms Company and the American Gas Association.[39] This work has resulted in the development of a three ton capacity prototype unit recently demonstrated to a number of appliance and engine manufacturers.

DESIGN AND OPERATIONAL FEATURES OF THE SIGMA GS-34 GASIFIER

Figures 4 and 5 show vertical and end cross section views of the SIGMA GS-34 gasifier. The cylindrical case serves as a central support for the engine and component parts. The compressor cylinders, carrying discharge valve plates and intake valve

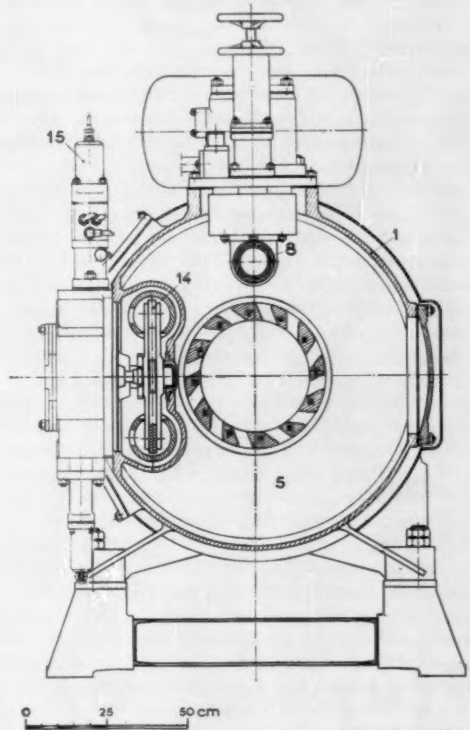
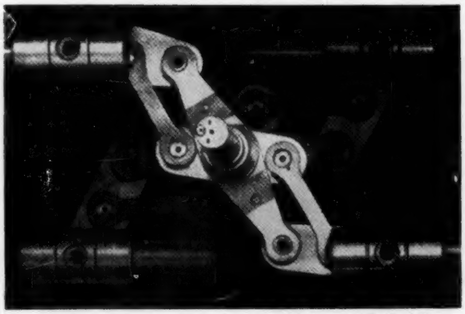


Figure 5 — Cross section end view of a SIGMA Model GS-34 gasifier.



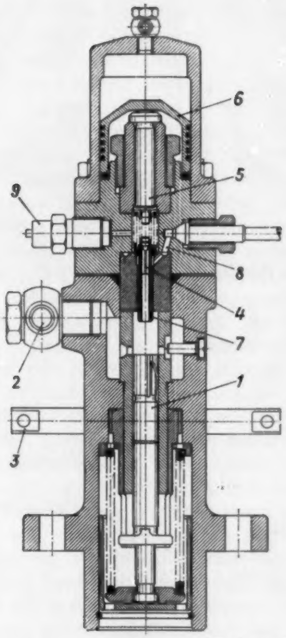
Courtesy of Professor G. Eichelberg, Swiss Federal Institute of Technology

Figure 6 — Synchronizing rods and linkages in their end positions.

support rings, are bolted on either end. Dished heads close the outer ends of the compressor cylinders to form the bounce chambers. The power cylinder assembly located in the center of the case is water cooled and consists of a scavenge cylinder and an exhaust cylinder bolted to a center cylinder section that carries the injectors and forms the combustion space. The scavenge cylinder has angular ports to create swirl of incoming air. Intake air enters the compressor cylinder through automatic reed type valves and discharges through automatic "Hoerbiger" type spring loaded valves into the scavenging air receiver formed by the space between the central case and the power cylinder. An exhaust collector assembly directs the exhaust gases out of the engine case to the exhaust manifold. The weight of the piston assemblies is transmitted to the cylinder surfaces by three wear bands installed on both power and compression pistons. The power pistons are fitted with 6 pinned compression rings and cooled by oil which enters through the center of the bounce covers and flows to and from the pistons by a system of sliding concentric tubes. The outside tube is fitted with sealing rings to minimize leakage of piston cooling oil into the bounce chamber. A balance pipe equalizes the pressures between the two bounce chambers and also serves to guide the pistons and keep them from rotating. While the major forces acting on the pistons are balanced by pneumatic pressures, a synchronizing mechanism as shown in Figure 6 is connected to the two compressor pistons to equalize any slight force differences due to friction, wear or compression pressures. The synchronizing gear is also used to actuate the fuel injector pump and to drive a mechanical force feed lubricator which distributes oil to lubricate the power and compressor cylinders. A parallelogram type linkage is used in the SIGMA GS-34 while a geared rack and a pinion type is used in the Muntz CS-75 and some of the other designs. Here it should be pointed out that the synchronizing mechanism does not in any way limit the stroke of the pistons either inward or outward. Piston stroke is a function of the pneumatic forces acting on the piston and is primarily controlled by the quantity of fuel burned and the pressure maintained in the bounce chambers. The control of bounce chamber pressure is done by means of a stabilizer which correlates bounce chamber pressure with discharge gas pressure or scavenging air pressure. Thus it will be seen that piston stroke and cyclic speed, as well as the position of the inner and outer dead points will vary with the loads imposed on the pistons, that is, the power output of the engine.

Fuel Injection System

In a free piston engine it is necessary to inject fuel at a time when the pistons are near the inner dead point and when there is little or no motion of the moving parts. Since there is no flywheel or crankshaft, the pistons accelerate rapidly away from their inner dead points, which reduces the time available for combustion. Therefore, the correct amount of fuel must be injected rapidly and at the proper time, consistent with variations in piston stroke and load. To accomplish this an accumulator type fuel pump illustrated in Figure 7 is used[26]. Essentially the pump meters and stores the fuel under pressure during the inward stroke of the piston. At the proper time, a poppet type valve is opened to distribute fuel to the injectors. Thus injection proceeds independently of the moving parts and duration of injection is determined primarily by the pressure in the accumulator, the quantity of fuel injected and the area of the nozzle orifices. Referring to Figure 7, the amount of fuel injected is controlled by a helix cut in the pump plunger (1) whose rotation is controlled by a control rack (3). On the inward stroke of the power pistons, the parallelogram synchronizing linkage through a cam and timing lever linkage, lifts the pump plunger which meters the desired amount of fuel through a small non return valve (4), located in the center of delivery valve (7), into a chamber below the accumulator plunger (5) which is loaded by the compressed air piston (6) to give the desired injection pressure. Shortly before the end of the stroke, the continued upward travel of the pump plunger (1) lifts the spring loaded delivery valve (7) and the fuel in the accumulator is delivered through one or more pipes (8) to the injectors. Injection timing may be adjusted by a servomotor which acts on the lever connected to the synchronizing mechanism. The earlier design GS-34 units are equipped with two fuel injection pumps. At full load the larger pump delivers some 95% of the fuel charge at a pressure of around 6000 psi to four injectors that spray directly into the combustion chamber. The smaller pump delivers the remaining 5% of the



Courtesy of Professor G. Eichelberg, Swiss Federal Institute of Technology

Figure 7 — Fuel accumulator pump.

fuel at a lower pressure to pre-combustion chamber nozzles located 180° apart. The pre-combustion chamber outlets are so arranged to produce additional swirl in the combustion zone to break up the fuel sprayed by the direct injectors. This arrangement provides good combustion characteristics and gives an invisible exhaust up to maximum load. Similar results are obtained in later designs with a single pump feeding all six injectors.

Controls

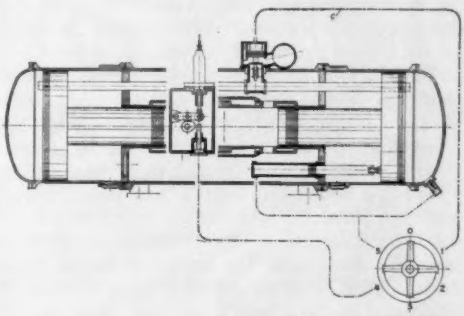
The control of free piston engines[5, 11, 13, 22, 23, 26, 27, 28, 29] primarily involves: (1) a means to start and stop the engine, (2) regulation of piston motion for stable operation or bounce control and (3) adjustment of power output to suit load requirements.

Starting

Figure 8 illustrates the starting system used in the GS-34[26]. To prepare the engine for starting, the control wheel is placed in position 5. This admits high pressure air from the "starting bottle" (tank) to a small air cylinder that moves the pistons to their outer dead position. At the same time the air from the bounce chamber is discharged by a relief valve actuated by the starting air pressure. The air supply is shut off at position 0. In position 1 a predetermined amount of air is admitted to the starting air vessel on the engine through a spring-loaded non-return valve that prevents the air from entering the bounce chamber or leaving the starting vessel. Starting takes place by moving the handle to position 2. This closes the air bottle and vents the starting air supply pipe to the atmosphere. The starting valve then opens and air rushes through the balance pipe to both bounce chambers forcing the pistons rapidly together. As the pistons approach the center of the cylinder, the motion of the synchronizing linkage actuates the fuel pump, for subsequent injection and combustion. Since there is no flywheel effect, the engine must start on the first inward stroke otherwise the starting procedure must be repeated. By the same token the engine will stop immediately if a misfiring condition develops. Normally the amount of fuel injected for starting corresponds to that required for average length of stroke. Continuing the handle to position 3 closes the air supply lines which in turn closes the starting air valve and prevents cushion air from entering the starting vessel while the engine is running. When the handle is moved to position 4, the plunger of the fuel pump is held up by compressed air and the engine stops.

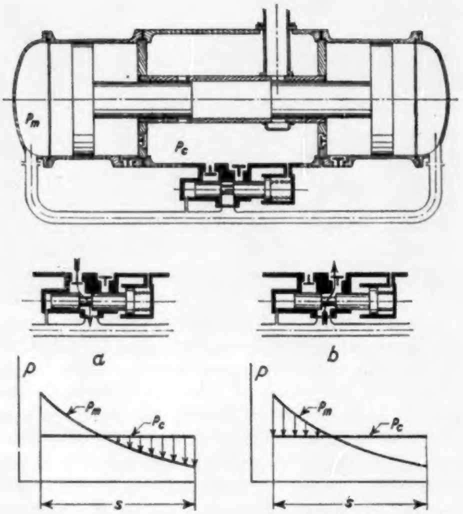
Bounce Chamber Control and Stabilizer Action

When combustion occurs on the starting stroke of a free piston engine, a set of controls must come into play immediately to control the motion of the pistons and govern the output of the engine. The flow characteristics of the commonly used axial flow power turbine connected to the gasifier are similar to an orifice of almost fixed area. Therefore any increase in power output and gas flow is accompanied by an increase in turbine inlet pressure which means the gasifier delivery pressure and the scavenging air pressure must increase accordingly. This, in turn, increases the pneumatic pressures



Courtesy of Professor G. Eichelberg, Swiss Federal Institute of Technology

Figure 8 — Schematic diagram of starting system on a SIGMA Model GS-34 gasifier.



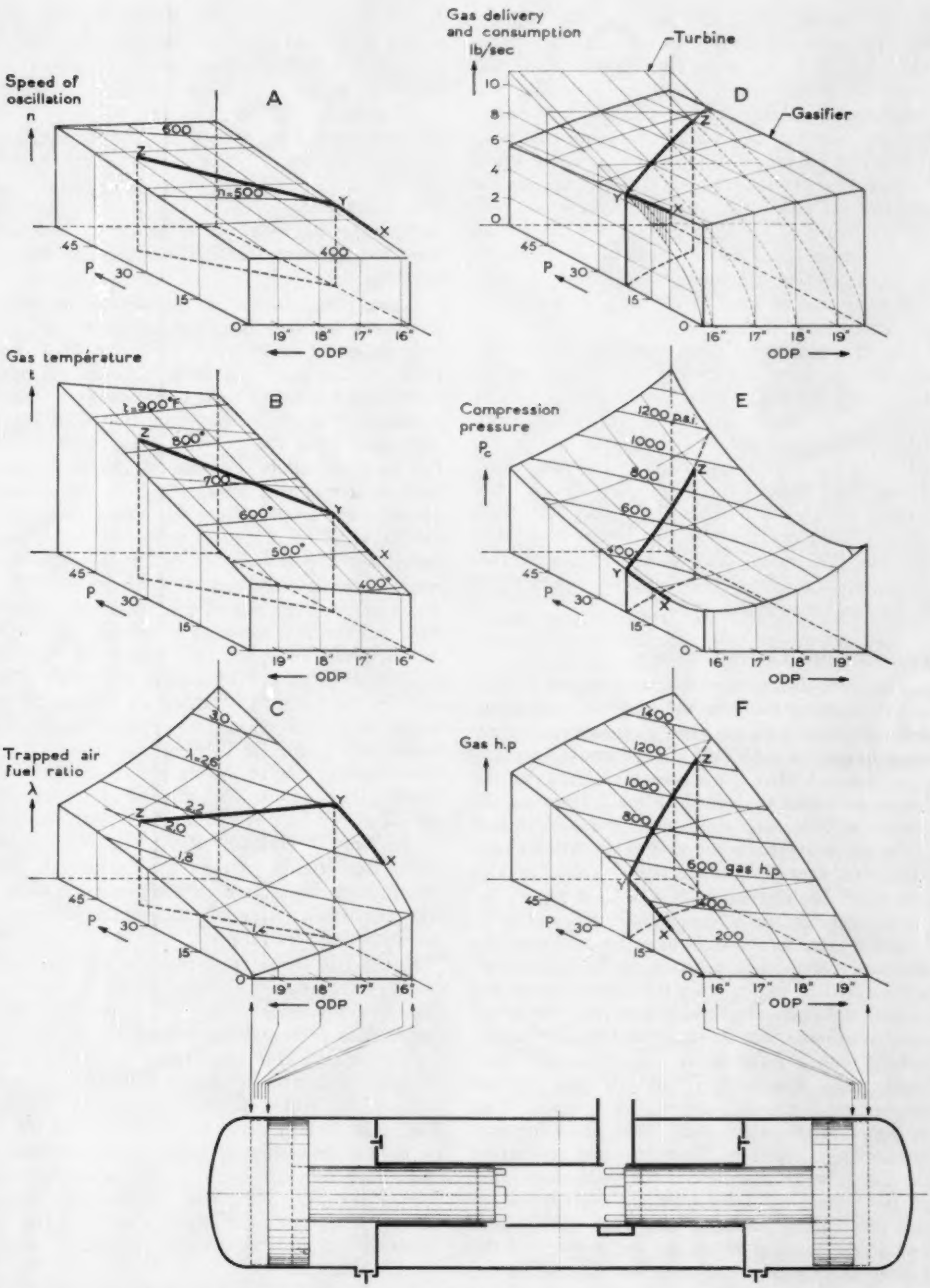
Courtesy of R. Huber, SEME

Figure 9 — Diagrams illustrating operation of cushion control or bounce chamber.

acting on the pistons. The motion of the pistons in a free piston engine may be compared with a mass oscillating between a system of springs with variable spring rates. Here the speed and stroke of the pistons depends on their mass or inertia, frictional forces and the combination of pneumatic spring forces imposed by the power cylinder, the compressor cylinder and the bounce chamber. Thus the length of the piston strokes and the position they come to rest on both the outward and inward strokes will vary depending on power output. The power output and the length of the outward power stroke is primarily controlled by the amount of fuel burned. This also determines the length of the suction stroke of the compressor and the position of the outer dead point which in turn is proportional to the amount of air delivered per cycle. The energy built up in the bounce chamber controls the length of the inward stroke and the inner dead point, also the ODP. This also determines the pressure of air delivered by the compressor and the compression pressure in the diesel power cylinder. When the amount of fuel injected increases, the air pressure in the bounce chamber must also increase to stop the pistons and return them to their proper inward position: That is, to maintain sufficient but not excessive compression of the combustion air at all operating conditions. The control of bounce chamber air pressure is accomplished by a stabilizer as illustrated by the schematic diagram shown in Figure 9. In effect, the stabilizer correlates the pressure in the bounce chamber with scavenging air pressure and gas delivery pressure consistent with the fuel pump rack setting. Basically, the stabilizer consists of a spring loaded differential slide valve which allows the transfer of air between the air box and the bounce chamber. The differential piston is so designed that it is always possible to transfer air from the engine case to the bounce cushions during one part of the stroke and conversely to return air from the bounce chambers to the engine case during the other part of the stroke. For any particular power output the mean pressure in the engine case is fairly constant. Therefore it will be seen from the diagram in Figure 9 that an increase in engine case pressure will increase the amount of air transferred to the bounce chambers. Conversely as the engine case pressure decreases with lower fuel rack settings, the amount of air transferred to the bounce chambers decreases. Thus for each fuel rack setting the stabilizer automatically adjusts bounce chamber pressure to control piston motion within optimum limits for satisfactory operation. In practice, it is also necessary to control both the upper and lower limits of piston stroke. The lower limit is set by the necessity to uncover the intake and exhaust ports sufficiently to permit adequate scavenging. The upper stroke limit is set so that mechanical contact will not occur at the end of the stroke or that excessive compression pressures will not be encountered on the inward stroke. To accomplish this, the fuel rack is equipped with maximum and minimum stops to control the range of fuel delivery.

Load Control

The method of load control will differ according to the particular application but power variation is primarily obtained by controlling the fuel delivery to the gasifier. An increase in fuel supply will lengthen the piston stroke and increase the mass flow, pressure and temperature of the gas. The fuel supply may be controlled either automatically by a governor fitted to the turbine (as for electrical generating sets), or manually controlled in the case of locomotives, ships or automotive applications. For starting, idling and low load conditions the mass flow of gas produced by the gasifier may exceed the "swallowing capacity" of the gas turbine. Under such conditions it becomes necessary to discharge part of the gas to the atmosphere. This is accomplished by a blow-off valve installed between the gasifier and the turbine inlet which may be controlled by a hydraulic servomotor operated by the turbine governor or manually controlled. In multi-unit systems, one or more gasifiers may be shut down when load demand decreases sufficiently. Here, automatic shutoff valves are usually interposed in the gasifier discharge line to isolate a particular gasifier from a common manifold during shutdown. Marine vessels are usually equipped with



Courtesy of R. Huber, SEME

Figure 10 — Influence of piston stroke and delivery pressure on operating characteristics of a SIGMA Model GS 34 gasifier feeding a gas turbine.

turbines having ahead and astern turbine wheels. Here the turbine is equipped with a three way valve which will direct gas to the ahead wheel for normal cruise or to the astern wheel for reversing and maneuvering. During starting and low load operation the valve may be set at some intermediate position so that the torque produced by the ahead wheel balances that of the astern wheel. This increases the effective gas flow capacity of the turbine and provides a means of handling gasifier flow rates from zero propeller speed to full speed ahead or astern. In locomotive or automotive applications a three way valve may be used to discharge excess gas to the atmosphere during start-up, idling and low loads. However, since these type applications usually operate under idle or low load conditions for appreciable periods, a recirculation process may be employed to improve part load fuel consumption. Here recirculation involves by-passing part of the air from the engine case back to the compressor intake. This reduces the work done by the compressor cylinder and heats the intake air which permits stable operation at low compression ratios and low flow rates. With this arrangement, fuel consumption at idling can be reduced to some 8%[28] of the fuel consumption at full load.

Performance Characteristics

The three-dimensional diagrams shown in Figure 10 illustrate the influence of piston stroke and delivery pressure on the various characteristic operating factors of a SIGMA GS-34 gasifier feeding a gas turbine[5]. Here it has been considered that the length of stroke as defined by the position of the pistons at their outer dead point and the pressure of the gas delivered by the gasifier are the two basic parameters that characterize the operation of this type unit[5]. In diagrams A, B, and C, of Figure 10, piston stroke increases from right to left and in D, E, and F from left to right to correspond with the illustration of the free piston gasifier. Piston stroke is measured in inches from the center line of the gasifier and delivery pressure is in psig. As previously mentioned, when a free piston gasifier feeds a turbine with a given nozzle area, there is a particular mass flow of gas that can pass through the turbine for each turbine inlet pressure. This in turn determines the outer dead point for each turbine inlet pressure. Then the law connecting the gas delivery pressure and the outer dead point of the pistons is defined by the intersection of the two curved surfaces in Figure 10D where one of the surfaces represents the delivery of the gasifier and the other the mass flow of gas that can pass through the turbine. The projection of this curve on the horizontal plane gives the relationship between the gas pressure and the position of the outer dead points under steady running conditions. It will be seen that at point X, which corresponds to zero or no load condition, the gasifier delivery is greater than turbine consumption. Therefore, the excess gas produced (indicated by the cross hatched area) must be allowed to escape. At point Y the delivery is equal to consumption and the discharge of excess gas ceases. Point Z is at a gas delivery pressure of 43 psig which corresponds to full load. This relationship for X, Y, and Z is the same for each of the diagrams where the vertical ordinates show the different characteristics of a gasifier-turbine unit.

Figure 10A indicates that the speed of oscillation increases with gas delivery pressure but is practically independent of the length of the stroke. This comes from the fact that at a given pressure the mean piston speed increases in the same ratio as the length of stroke. The gas delivery temperature in Figure 10B varies from 478°F. when idling, to 842°F. at full load. It will be seen that discharge temperatures always remain sufficiently high to minimize any corrosion problem from sulfur containing acids. On the other hand the gas temperatures are sufficiently low to permit the use of less expensive non-critical materials for turbine construction and to minimize any corrosive effects from vanadium or other ash forming materials in residual fuels. The ratio λ given in Figure 10C, represents excess of trapped air or excess combustion air, which is defined by the weight of trapped air divided by the weight of air theoretically required for combustion, shows that under all running conditions, except when starting, the excess air is at least 100%. It should also be pointed out that the excess scavenging air at full load is 150%. As a result the total air delivered at full load is about 5 times the weight of air theoretically required for combustion. This feature together with high compression pressures and temperatures materially aids in the burning of heavy residual type fuels.

Figure 10E shows the increase in diesel cylinder compression pressure when the stroke increases and the delivery pressure rises. It will be seen that diesel compression pressure is not proportional to the scavenging pressure as in a crankshaft engine but it can be chosen within limits for each delivery pressure by adjusting the return energy of the bounce chamber. The return energy depends on the level of the air pressure in the cushion cylinders which is controlled and corrected by the stabilizer. This is another basic feature peculiar to the free piston engine which permits it to operate at high excess air ratios yet maintain engine compression pressures within operating limits.

The diagram of gas horsepower in Figure 10F shows that the effect of an increase in stroke is greater with higher gas-delivery pressures. Also the same gas horsepower can be obtained with a

short stroke and a high gas pressure, or with a long stroke at a lower gas pressure. Here gas horsepower has been defined[11] as the power available from adiabatic expansion of the gas down to atmospheric pressure. In other words, it is the power that would be obtained if the gas was expanded in a 100% efficient turbine exhausting to the atmosphere. Gas horsepower is then defined as the product of adiabatic heat drop and the mass flow of gas which may be expressed by the following equation:

$$\text{Gas hp.} = 1.417\, m\, C_p\, T_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right)$$

where

m = mass flow from gasifier, #/sec

Cp = specific heat of gas at constant pressure, BTU/#/°F.

T_1 = gas delivery temperature, °R

P_1 = gas delivery pressure, psia

P_2 = gas atmospheric pressure, psia

n = ratio of specific heats, $\frac{C_p}{C_v}$

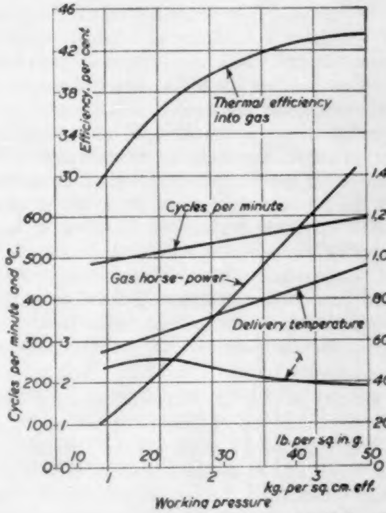
Thus it will be seen that power output is primarily a function of the mass flow of gas, temperature and discharge pressure.

Characteristic performance data for a single GS-34 gasifier are also shown in Figure 11. Here it will be seen that as gas hp increases there is a corresponding increase in thermal efficiency to about 43% at rated load of 1250 gas hp and 43 psig discharge pressure with even further increases in efficiency indicated for higher power outputs. The curve λ, which is the ratio of the weight of air trapped in the cylinder divided by the weight of air theoretically required for complete combustion is decreasing with increased power output; however, with the large excess of scavenging air, combusion is satisfactory. In fact, overload tests have been carried out at 51 psig discharge pressure and 1530 gas hp with the exhaust reported as practically invisible.[11] As further increase is indicated before the smoke limit is reached, tests are being carried out with the object of increasing rated power output. Whereas efficiency and fuel economy fall off rather rapidly as load is decreased, the use of recirculation as discussed in the previous section and the possible incorporation of compressor intake air throttling are indicated to minimize this problem.

LUBRICATION

The GS-34 gasifier is equipped with two separate lubricating oil systems similar to that used in large low speed diesel engines: (1) a circulating oil system to cool the power pistons and lubricate the synchronizing rod mechanism and (2) a force feed system for lubrication of the compressor cylinders and the power cylinder. The piston cooling oil system includes an oil sump, oil cooler, motor driven pump, strainer and pressure relief valve. The pump circulates from 80 to 100 gallons of oil per minute to the pistons at a pressure of 30 to 40 psig. As illustrated in Figure 4 oil enters the engine through the center of the bounce chamber covers and is fed to the piston crown by a system of concentric sliding pipes. An oil baffle distributes the oil in the piston to provide maximum cooling of the piston crown and ring band area. The hot oil from the piston then returns through the spaces between the sliding concentric pipes to the oil sump where it is cooled and recirculated. Part of the piston cooling oil is also used to lubricate the synchronizing mechanism. In order to minimize leakage of the piston cooling oil, the outer concentric oil tube is fitted with a ring type seal. Similar seals are also installed on the synchronizing rods and in the compressor pistons where they slide on the balance tube. A low viscosity oil is desirable to obtain good transfer of heat from the piston, yet the viscosity must be sufficiently high to provide satisfactory lubrication of the synchronizing mechanism and the various tubes and rod surfaces in contact with the ring seals. As a compromise between these two requirements an SAE-20 grade oil is normally used. With the amount of oil circulated, discharge oil temperatures are normally maintained around 145°F. at full load to minimize deposit formation in the piston crown. Any increase in discharge temperature is usually an indication of poor ring condition or some abnormal operating condition.

Oil is metered and distributed to both the compressor cylinders and power cylinders by a multi-



Courtesy of R. Huber, SEME

Figure 11 — Performance curves of a SIGMA Model GS-34 gasifier plotted against working pressure.

barrel force feed lubricator that is driven by the synchronizing linkage through a cam and ratchet mechanism. The compressor cylinders have eight plain lubricator ports equally spaced around the circumference. The power cylinders have two V channeled lubrication points, 180° apart, located outboard of the intake and exhaust ports, adjacent to the piston ring belt area at the outer dead point, and from 3 to 6 equally spaced V groove lubrication points inboard of both intake and exhaust ports. Non-return valves are normally placed near the point of oil delivery to minimize the effect of cylinder gas pressures on the flow of lubricant. An SAE 40 or 50 grade oil is normally used to lubricate the cylinders of the SIGMA GS-34 gasifier while an SAE-20 grade is used in the Muntz CS-75 for both piston cooling and cylinder lubrication.

During early development work it became evident that the lubrication requirements of a free piston engine were similar to that of a high output diesel engine but somewhat more demanding because of the high pressures and temperatures encountered in the power cylinders. The requirements are further amplified as the oil used for the power cylinders should also be a good air compressor lubricant and minimize deposit formation on compressor discharge valves. The deposit forming characteristics of the piston cooling oil also assume major importance since any leakage of this oil around the various seals ultimately finds its way into the air box together with any excess oil from the compressor cylinders. As power output increases, the scavenging air temperature also increases resulting in air box temperatures in the range of 425 to 465°F. at full load. Therefore any oil reaching the air box will have a tendency to form deposits at these temperatures which can lead to air box fires. While an air box fire has not resulted in an explosion risk or engine damage, it will cause engine stoppage by reducing the oxygen content of the combustion air. It should also be mentioned that the lubricator lines to the power cylinder must pass through the scavenging case and the cylinder lubricant should not form deposits in the tubes or check valves during the period it is subjected to the high air box temperatures. In addition to minimizing wear and deposits the cylinder lubricant must also protect the bronze or aluminum wear bands, installed on the compressor and power cylinders, against corrosion.

The severity of these requirements and the reliability of the engine will of course depend on engine design, the selection of materials for the rubbing surfaces, finish and fit of the parts, the location of lubrication points and the rate of oil delivery, the degree of break-in, the manner in which the engine is installed and operated as well as the use of the proper type of oils. In the development of free piston engines the major problems encountered from a lubrication standpoint have been:

1. Air box fires from deposits built up on compressor discharge valves and deposits in the scavenging air box.
2. Piston ring groove deposits.
3. Ring wear, scuffing and breakage on both power and compressor pistons.
4. Cylinder liner wear.

Here the continued build-up of deposits on the compressor discharge valves will tend to hold the valves open and increase air temperatures which in turn increases deposit build-up in the air box. As the deposits become thicker and hotter, then they may ignite spontaneously or as a result of sparks from the scavenging ports.

Considerable efforts have been made both here and abroad to improve engine design so that both thermal stresses and pressure stresses have been reduced as much as possible. As continued efforts are made to increase the power output from the engines, the provisions for adequate cooling and control of combustion pressures will become increasingly important. The use of water cooled compressor head plates has helped reduce discharge air temperature to a more reasonable level to reduce deposit forming tendencies on the discharge valves. Elimination of hot spots in the scavenging air box by better cooling has also reduced the tendency toward deposit buildup and air box fires. Improving the hardness and surface finish of the sliding pipes and rods has permitted better sealing to reduce the amount of piston cooling oil leakage contributing to air box deposits. Similarly a reduction has been made in the amount of oil fed to the compressor cylinders and the sealing rings on the power piston trunks.

Improved piston cooling has minimized the problem of ring groove deposits and ring breakage, however, scuffing and wear of piston rings and cylinders continue to be a major problem. The use of taper-faced ferrox filled piston rings in conjunction with "lubrited" cylinder surfaces finished to specifications of diesel engine practice and the use of a Series II type oil in the GM-14 gasifier is reported to have eliminated scuffing and greatly reduced ring wear in extended laboratory tests burning heavy residual fuel.[1] Further experience will be required to determine the extent this is reflected in actual practice. Three different kinds of scuffing have been encountered with the GS-34 and CS-75[34]: (1) that caused by "swarf" or metal chips removed when running in a new engine, (2) that caused by nonrectilinear motion of the moving parts, and (3) that caused by maldistribution of the oil on the sliding surfaces. The elimination of "swarf" has so far proven difficult in new engine cases but a routine of very short break-in runs is reported to eliminate

this problem before it causes damage. Reactions of the synchronizing gear and slightly wavy cylinder surfaces have also been suspected as a cause for occasional scuffing. In addition to adequate machining, it may be desirable to further study break-in procedures including the possible use of special break-in oils to develop suitable running surfaces to minimize this problem. Adequate break-in also determines to some extent the wear rate that will be encountered later. In addition to proper break-in and proper machining and finish of the parts, improved oil distribution is considered a major factor in reducing scuffing and wear. Reported improvement[34] has been obtained by: (1) increasing the number of oil feed points in both compressor and power cylinders, (2) increasing the frequency of lubricator oil pulses from one pulse per 200 piston strokes to one pulse per 20 strokes in the GS-34, in the CS-75 from one pulse per 200 strokes to one pulse per 6 strokes. A substantial reduction in over-all oil consumption has also been effected by reducing the volume per pulse. Ring wear has been reported[34] as low as 0.007 inches per thousand hours, based on radial measurements, which would correspond to a replacement period of 1500 to 2000 hours. Although it has been demonstrated that the rings on a GS-34 gasifier can be changed out in a matter of 40 minutes, further reduction in ring wear is desirable. The following figures have been reported[34] for power cylinder liner wear for GS-34 gasifiers which may vary as follows depending on the type fuel burned:

Marine Diesel Fuel — 0.002-0.004 in./1000 hrs.
Bunker "C" Fuel — 0.005-0.007 in./1000 hrs.
Compressor cylinder wear is relatively smaller.

A great deal of development work has and is being carried out to provide improved lubricants for the free piston engines. At the present time, the type oils used have gradually progressed from a straight mineral oil initially used for both piston cooling (SAE-20) and cylinder lubrication (SAE-50) to additive type oils in the Series II and III additive levels for cylinder lubrication (SAE 40 or 50) and a Series I level of the SAE-20 grade for piston cooling. The Muntz CS-75 type is reported[34] to give encouraging results with a Series II SAE-20 oil for both piston cooling and cylinder lubrication. These higher additive oils are considered desirable in reducing the incidence of air box fires as well as reducing scuffing and wear. As further advances are made to increase power output, it is expected that even further improvements in lubricants will be required. This will naturally require close cooperation between the engine builder and the oil supplier as further steps to increase the demands on lubricating oil preformance could have a tendency to reduce the reliability of the machine and increase operating costs.

In addition to the effect of engine design, manufacture and lubricating oil development, the manner in which the engine is installed and maintained has an important bearing on these problems of air box fires, scuffing and wear. For example, at full load approximately 8.5 pounds of air per second passes through a GS-34 gasifier. If this air is not adequately filtered before entering the engine, any air borne dirt will not only increase wear of piston rings and cylinder liners, but it may also contribute to the build-up of deposits on compressor valves and in the air box. In such cases the cylinder oil will have a tendency to act as a binder for the air borne dirt particularly if feed rates are excessive which may lead to excessive air box and compressor discharge valve deposits and eventual air box fires regardless of the superiority of the lubricating oil used. Difficulties would also be expected if the operator or control system permitted excessive overload for extended periods. Thus the manufacturer can make the best engine in the world and the petroleum supplier can develop the best lubricants but if the engine is not installed and operated properly these developments are of no avail.

The lubrication requirements of the gas turbine and gear boxes that may be connected to a free piston gasifier do not normally present any unusual problems and conventional rust and oxidation inhibited turbine grade oils are usually satisfactory. In many cases it is desirable to use the same oil for both gas turbine bearings and reduction gears. In such installations the requirements of the reduction gears determines the viscosity of the oil used.

FUEL

As previously indicated, the over-all efficiency of present free piston-gas turbine units either approaches that of supercharged diesel engines or is slightly less. Therefore in order for a free piston to be competitive it must burn a lower cost residual or Bunker C type fuel. As the free piston engine may operate at compression ratios up to 50 to 1 and with high excess air ratios, it is possible to burn a wide variety of fuels ranging from gasoline to heavy residual or Bunker C type fuel. The main limitations in burning the heavier fuels are (1) the ability to heat the fuel to a high enough temperature that it can be pumped and injected, consistent with limitations that may be imposed by the fuel pumps and injector nozzles. This would include the ability of these parts to operate properly at the fuel heating temperatures imposed, including possible problems due to leakage or build-up of fuel deposits; (2) the ability to burn the fuel at acceptable fuel consumption rates without encountering excessive wear of the moving parts or down time.

The first of these considerations is largely a matter of fuel system design and proper installation of the

fuel system to handle heavy fuels. In most free piston engine heavy fuel installations, provisions are normally made to start, stop and maneuver the engine on light diesel fuel and to burn heavy fuel at the higher loads. Suitable heating equipment and a centrifuge to remove water and abrasive solids are the major pieces of equipment used for treating the fuel. Since the exhaust gases from the gasifier enter the turbine at approximately 840°F., there is no problem of turbine blade corrosion or deposit build-up from sodium, vanadium or other ash forming materials in the heavy fuels. Accordingly, the fuels are not normally treated to remove or inhibit these materials as would be the case where such fuels are burned in a conventional gas turbine. In gas turbines, the gas inlet temperatures may be in a temperature range of 1250 to 1500°F. where corrosion can occur.

The second consideration, namely the ability to burn the heavy fuel at acceptable consumption rates is also a function of the design of the injection equipment and proper timing. Since the scavenging air temperatures in a free piston engine run around 400 to 440°F. and well above the dew point of sulfur-containing acids, corrosive wear with high sulfur residual fuels is minimized as compared with some diesel engines. However, the abrasive wear rates from ash forming materials that may be present in these fuels may be as high or higher than encountered in conventional diesel engines because of the higher combustion temperatures and pressures. Here the use of specially compounded additive type lubricants in a suitable viscosity range may help alleviate this type wear. Further experience may also indicate the desirability of developing suitable treating procedures for certain fuels to minimize the abrasive effects of ash forming materials.

GM Research has reported on results obtained in their GM-14 engine using a Bunker C type fuel normally used for plant heating purposes.[1] The fuel used averaged 3.5% sulfur content with some deliveries running as high as 4.2%. API gravity was in the range of 10.9 and the viscosity was 149 seconds Saybolt Furol at 122°F. or around 3400 seconds Saybolt Universal at 100°F. The first test was run at 1250 gas hp for 1728 hours after which several modifications were made and a second endurance test of 4500 hours duration was carried out with over 90% availability of the engine. Since this time, it is understood that General Motors have accumulated over 12,000 hours test experience using heavy fuel which indicates the engine is capable of running on such a diet. While specific wear figures are not available for these latter tests it is understood they were in an acceptable range with no evidence of scuffing. As a result of these tests further ring changes are indicated to improve top ring performance.

In this connection the top ring on the exhaust piston normally shows the most wear since it is subjected to the highest temperatures. The following wear rates have been reported with a SIGMA GS-34 gasifier installed in the coasting vessel "Cantenac" after some 7200 hours at sea operating on a Bunker C type fuel with the viscosity ranging from 2500 to 3600 seconds Saybolt Universal at 100°F.[30]

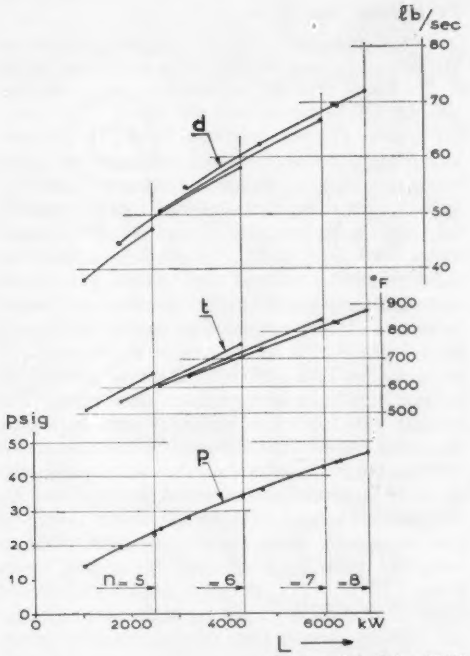
Part	Wear per 1000 hrs.-inches
Exhaust cylinder liner	0.0032
Scavenge cylinder liner	0.0027
First diesel piston ring-exhaust end	0.0107
First diesel piston ring-scavenge end	0.0063
Wear on other piston rings	0.0032
Wear on compressor cylinder	0.0020
Wear on compressor piston rings	0.0032

FREE PISTON APPLICATIONS

In March 1958, over 350 gasifiers of the SIGMA-GS-34 type were reported in service or under construction by the various licensees.[5] Of these, over 90 gasifiers were in commercial operation as of October 1957, which had accumulated a total running time of some 250,000 hours. So far as it is known, all of the commercial free piston gasifier installations in service today stem from the original Pescara patents although there have been several other interesting experimental designs. At the present time, the commercial application for the GS-34 type free piston-gas turbine sets have included electrical power generation, ship propulsion and auxiliary generating sets, locomotives and pumping service with shaft outputs ranging from 1000 to 8600 hp.

Stationary Electrical Power Generation

As of June 1, 1956[29] five 600 kw sets had completed 35,500 accumulated hours of operation with an average of 7000 hours per set. In these units a single GS-34 gasifier drives a 4 stage, 11,700 rpm turbine which is geared down to 1500 rpm through a single stage gear reducer. One of the turbines from this group was inspected after 12,000 hours and was found in excellent condition. A 1500 kw set has been in operation on the French Electricity network for over 7000 hours. Sets of this type have two GS-34 gasifiers connected to one four stage, 9000 rpm turbine which drives an alternator through a 6 to 1 single stage reduction gear. Specific fuel consumption has been reported as 0.56 #/kw-hr. at the terminals. In the 1500 kw sets, the admission casing of the turbine may be divided to permit operation on one gasifier. Here one gasifier can be stopped for maintenance while 600 kw may be obtained with a single gasifier.



Courtesy of R. Huber, SEME

Figure 12 — Performance tests of eight SIGMA Model GS-34 gasifiers feeding one gas turbine driving a 6000 KW generator.

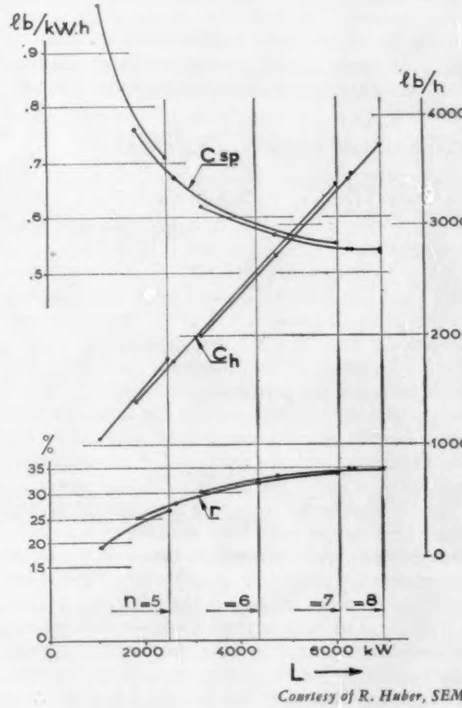
The largest stationary set in operation is a 6000 kw unit at Cherbourg, France which consists of 8 GS-34 gasifiers supplying one 9 stage power turbine direct connected to a 3000 rpm, 6000 kw alternator. In this plant it has been demonstrated that one of the gasifiers can be taken off the line for inspection or repairs while the remainder will carry the 6000 kw rated load. The results of official tests on this unit have been reported[5] and a set of the performance curves is shown in Figures 12 and 13. It will be noted that the maximum available power with eight gasifiers in operation is 6900 kw. At the 6000 kw rated load, overall specific fuel consumption is indicated to be 0.545 #/kw-hr. at the terminals with an overall thermal efficiency of 34.5%. The separate curves for each performance characteristic illustrate the effect of decreasing the number of gasifiers from 8 to 5.

The largest stationary installation on order is a 36,000 kw plant intended for a power station in Singapore. It is to be composed of 6 sets similar to the Cherbourg station, each consisting of 8 gasifiers connected to a single turbine.

Marine Sets

As previously mentioned, the French Navy has been interested in the development of free-piston engine machinery since the beginning of 1938. A prototype mine sweeper was ordered in 1951 and (in the spring of 1953) was the first free piston ship at sea. Twenty-one of these French mine sweepers are now in operation and the gasifiers have accumulated 40,000 total running hours as of June 1, 1956. Each mine sweeper has two propellers each driven through reduction gearing by a gasifier-turbine unit. The turbines have ahead and astern wheels in the same casing. To stop the propeller shaft, the ahead and astern wheels are supplied with gas simultaneously. Fuel consumption on one of these mine sweepers has been reported[30] as 0.430 lbs./shaft hp. hr. This includes a gear reduction efficiency of 96% and a turbine efficiency of 84% including the reversing drive.

Two small cargo vessels of 1100 tons each, were placed in operation in 1953. Each vessel is equipped with two free piston gasifiers and two 1000 hp turbines connected to a common reduction gear and a single output shaft. Over 25,000 hours of operation have been recorded[30] for the 4 gasifiers and apart from 3 weeks annual stop for ships repair and general maintenance, they have had only 8 and 4 days



Courtesy of R. Huber, SEME

Figure 13 — Performance tests of eight SIGMA Model GS-34 gasifiers feeding one gas turbine driving a 6000 KW.

interruption of duty for free piston machinery trouble.

As mentioned earlier, a 6000 hp free piston gas turbine set composed of 6 gasifiers and two 3000 hp turbines was installed by the Cleveland Diesel Engine Division of General Motors in the converted Liberty Ship "William Patterson". Official sea trials were completed in September 1957 and the vessel sailed on its maiden voyage to Europe in October 1957. Untreated Bunker C type fuel is being used and the operating experience is being followed with interest. Several other marine propulsion applications are in the planning stage or under construction. Of these, it is reported that two 16,000 shaft hp sets are under construction for frigates for the French Marine Nationale.[34] Each set is to have two propeller shafts driven by two double turbines supplied by eight GS-34 gasifiers.

Locomotive Application

A prototype free piston locomotive was constructed by la Regie Nationale de Usines Renault and put into service in 1953.[5] This unit has a single GS-34 gasifier connected to a 1000 hp, 12000 rpm turbine.[31] A 6 to 1 reduction gear and a reversing gear box transmits turbine shaft power to the driving wheels. This locomotive has been operated over 6000 hours or 150,000 miles in passenger service on the French National Railways. On the basis of this experience, it is understood two larger locomotives of 2000 to 2400 hp output are under construction.[31]

Pumping and Compressor Drives

The only pump drive unit reported so far is a single GS-34 turbine set installed near Paris in 1955 to pump water for the city. A large compressor plant composed of 15 GS-34 gasifiers is to be installed in the No. 3 Olefines plant of Imperial Chemical Industries (Ltd.) in Wilton, England some time this year. In this plant 13 gasifiers will drive four turbo-compressor sets and the other two gasifiers will drive a 1500 kw generating set. Pumping units have also been proposed for petroleum pipe line use. While the present units appear suitable for liquid product lines, further development will be required to run the gasifier on gaseous fuels and it is understood such work is under way by SIGMA and others.

In addition to the foregoing installations, several other applications have been studied or are under consideration. As discussed in the history section, experimental automotive applications have been made by General Motors in their XP-500 experimental car, and Ford in their Typhoon farm tractor. Baldwin-Lima-Hamilton have also proposed a small size gasifier set for various automotive applications. In addition, Renault[35] has proposed a truck composed of two 120 hp gasifiers feeding a single gas turbine.

Petroleum Industry

Several possible free piston engine applications have been proposed for use in the petroleum industry.[36] These include applications of both the SIGMA GS-34 gasifier and the Muntz CS-75 gasifier to drive (1) pumping units from 350 to 16000 hp. either alone or combined with electrical generator sets, (2) a 350 hp and up application for drilling units and (3) a combined free piston gasifier/steam turbo generator plant. In the latter application, six GS-34 gasifiers would drive a 4640 kw turbo-generating set and the exhaust gas, which contains approximately 80% unburned air, would be used as a highly preheated air supply to the main steam boiler.[11, 29] In addition to the above combined cycle, studies have included the use of gasifiers to exhaust direct to a high pressure steam boiler. The exhaust gas from the boiler is then expanded through a gas turbine connected to the main steam turbine. For a 25,000 kw plant is was figured that about 25% of the power would be developed by the gas turbine and 75% by the steam turbine to give an over-all plant efficiency of some 34% as compared with 28% efficiency for a pure steam plant. This would of course require a high pressure "Velox" type boiler to operate at the 40 to 45 psig gas delivery pressure from the gasifiers. Free piston engines have also been proposed as a source of high pressure, high temperature air for industrial purposes[32]. One application suggested is related to the rejuvenation of static oil fields by forcing high temperature — high pressure gas into them to fluidize the viscous oils trapped in the sands and make additional recovery of oil possible. Thus it will be seen there are several interesting applications for free piston engines and further studies will undoubtedly be made to find additional commercial applications.

FUTURE TRENDS AND OUTLOOK

While additional experience with various type fuels may be required to further establish the economic and reliability aspects of the free piston prime mover in different type applications, it must be remembered that the free piston engine is a relatively new development and further improvements can be anticipated. Following are several improvements that have been considered for free piston gasifiers:

Fuel System

Improvements in the fuel system to increase fuel economy by burning a larger amount of fuel over a shorter time interval to realize increased combustion efficiency have been considered. This is limited by the combustion pressures and temperatures that may be tolerated in present engine design. It has been pointed out that maximum limits for these values are approximately 1800 psi and 3200°F.[28]

Natural Gas

Developing the engine to operate on natural gas would greatly enlarge the scope of free piston applications to pipe line pumping and other petroleum and refinery uses.

Part Load Fuel Economy

Further improvements in recirculation and intake air throttling to permit lower fuel consumption and economy under idle conditions and low loads are desirable. This is considered particularly important to further extend the use of free piston engines to locomotive service and automotive applications where the engines operate at idle or part loads for appreciable periods of time.

Increasing Over-all Thermal Efficiency

It has been shown that the over-all efficiency of the present GS-34 gasifier-turbine at rated load is approximately 37½ percent at a *turbine pressure ratio* of 4 (44.1 psig turbine inlet pressure) and about 39.5% at present overload pressure ratio of 4.5 (51.5 turbine inlet pressure).[28] Here turbine pressure ratio is defined as

$$\frac{\text{turbine inlet pressure, psia}}{\text{turbine exhaust pressure, psia}}$$

The maximum over-all efficiency peaks at about 42% at a turbine pressure ratio of 6 which is considered as the best range for maximum over-all efficiency, maximum power to weight ratio, and the limit for turbine inlet temperatures that will still permit the use of non-critical materials for turbine construction, yet still provide a satisfactory fuel-air ratio. However, before a turbine pressure ratio of 6 can be achieved in practice, it will first be necessary to obtain satisfactory operation at a turbine pressure ratio of 5, corresponding to a turbine inlet pressure of about 58.5 psig and an over-all efficiency a little above 40% which is considered to be about the maximum limit for the power piston rings, scavenging air temperature, piston and cylinder cooling and lubrication of the present GS-34 design.[28]

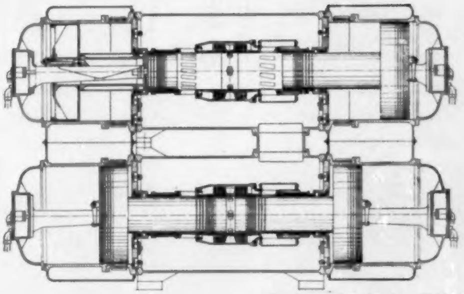
While it does not appear that an over-all thermal efficiency of 40% will be exceeded for some time, it is possible to practically double the power output of the same weight of the present free piston engine without using special construction.

Supercharging and Intercooling

While these methods appear attractive from a technical point of view, some question has been expressed whether the economic advantages would be so clear, particularly from an initial cost standpoint when the cost of turboblowers, heat exchangers and higher cost of turbines are considered.[5]

Afterburning

This has also been proposed[5, 28, 29] as a means to increase power output. Here it is indicated that a 30% increase in power output at an efficiency of about 32.2% may be obtained with the present GS-34 gasifier by burning additional fuel in a combustor installed between the gasifier and the turbine without resorting to expensive turbine construction. In this case turbine inlet temperatures would range below 1200°F. where vanadium corrosion of turbine blades begins to occur. By going to more expensive turbine designs, it is possible to further increase the power output in the ratio of the absolute temperature of gas after and before the combustor.



Courtesy of R. Huber, SEME

Figure 14 — Siamesed free-piston gasifiers.

Twinning or Siamesing

As illustrated in Figure 14 this also has been proposed[5, 26] as a means of doubling power output without appreciably increasing the space requirements. As previously discussed, this is the system employed by General Motors for their Hyprex free piston engine installed in the XP-500 experimental automobile and it will undoubtedly be considered for any future locomotive and other related applications where space requirements are at a premium. Essentially the system includes two gasifiers joined by a common air box and common intake and exhaust passages. In this manner it is possible to dephase the two power cylinders by a hydraulic dephasing device connecting the bounce chambers so that the adjacent pistons operate 180° out of phase.[26] Twinning also increases efficiency since the compressor pistons of adjacent cylinders pump air to scavenge the opposite power cylinder. This system also gives more uniform flow at the intake and tends to dampen out pulsations at the exhaust.

Size

Units of larger dimensions than the GS-34 gasifier have also been proposed.[5] Here a twinned gasifier with a 20″ bore as illustrated in Figure 14 has been considered as about the maximum size. The principal dimensions for this engine are shown in the following table:

Diesel cylinder bore	19.7 inches

Compressor cylinder bore	51.2 inches
Maximum possible stroke	31.5 inches
Stroke at full load	25.6 inches
Speed of oscillation	391 cycles/min.
Mean piston speed	1670 ft. per min.
Gas horsepower	4800 gas hp.
Power at turbine shaft	4000 shp.
Over-all length	20 ft.
Height	12.5 ft.
Width	35 long tons
Weight	6 ft.

In the development of this size gasifier, the three principal problems considered[5] are (1) the endurance of the combustion chamber walls, (2) guidance of the moving parts and, (3) the mechanical strength of the compressor pistons. None of these problems are considered insurmountable.

Finally, an entirely new scheme for the free piston engine may be found in the adoption of the free piston principle to generate electrical power by atomic energy. Here a sealed free piston atomic fuel powered engine might be visualized that could furnish electrical power to drive ships, aircraft, locomotives, stationary power plants and other electrical power applications.

SUMMARY

It is significant that a number of commercial free piston applications have been made within the last decade. Further advances in this field can be expected. The extent of future free piston engine applications will not only depend on their performance as compared with other prime movers but will also be compared on the basis of fuel requirements, first cost, maintenance costs, ease of overhaul, installation facilities, weight, space, shaft speed, torque characteristics and other considerations peculiar to this type power. It is also indicated that the development of improved lubricants will be required in future improved free piston engine designs which will require the close cooperation of the engine builders and the oil supplier as well as the operator who is developing practical experience with this type machinery.

BIBLIOGRAPHY

1. FLYNN, G. (JR.): "Observations on 25,000 Hours of Free Piston Engine Operations," SAE Transactions, Vol. 65, p. 508, 1957.
2. KASTNER, L. J.: "A Century in the History of the Reciprocating Internal Combustion Engine," The Institution of Mechanical Engineers Proceedings, Vol. 169, pp. 303-304, 1955.
3. NEUMAN, K.: "Junkers Free Piston Compressor," (in German) Zeitschrift Des Vereines Deutscher Ingenieure, Vol. 79, No. 6, pp. 155-160, Feb. 9, 1935.
4. "World Free-Piston Licensing Situation" Anon, The Oil Engine and Gas Turbine, pp. 241-2, Oct. 1957.
5. HUBER, R.: "Present State and Future Outlook of the Free Piston Engine," ASME Paper #58-GTP-9, presented at ASME meeting in Washington, D. C., March 2-6, 1958.
6. LONDON, A. L.: "Free-Piston and Turbine Compound Engine — Status of the Development," SAE Transactions, Vol. 62, pp. 426-436, 1954.
7. Free Piston Compressor," Anon, Mechanical Engineering, Vol. 67, pp. 852-854, Dec. 1945.
8. McMULLEN, J. J. & PAYNE, W. G.: "Performance of Free Piston Gas Generators," ASME Paper #53-S-18 presented at ASME Meeting, Columbus, Ohio, April 28-30, 1953.
9. TOUTANT, WILLIAM T. (LT.), USNR: "The Worthington —Junkers Free Piston Engine," Journal of the American Society of Naval Engineers, Vol. 64, p. 583, 1952.
10. FARMER, H. O.: "Free Piston Compressor Engines," Proceedings of Institution of Mechanical Engineers, Vol. 156, p. 253, Dec. 1947.
11. MUNTZ, F. A. I. & HUBER, R.: "The Free Piston Gas Generator Turbine as a Power Plant for Ship Propulsion," Institute of Marine Engineers Transactions, Vol. 66, ng, pp. 201-205, Sept. 1954.
12. "The Pratt and Whitney Aircraft Story," published by the Pratt and Whitney Aircraft Co., Hartford, Conn., 1950.
13. LASLEY, ROBERT A. & LEWIS, FRANK M.: "Development of High Output Free Piston Gas Generators," ASME Paper #53-S-34 presented April 30, 1953, Columbus, Ohio.
14. EHRAT, A. J.: "Free Piston — Gas Turbine Prime Movers — A Review of Basic Principles," ASME Paper #54-A-67, presented in New York, Dec. 3, 1954.
15. "Development of the Free Piston," published by Baldwin-Lima-Hamilton Corp., Hamilton, Ohio.
16. "New Free Piston Engine Unveiled," Anon, Automotive News, p. 70, Sept. 9, 1957.
17. McMULLEN, J. J. & RAMSEY, R. P.: "The Free Piston Type of Gas Turbine Plant and Applications, ASME Paper, Trans. ASME, Vol. 76, pp. 15-29, 1954.
18. CALDERWOOD, J.: "Some Researches in Internal Combustion Engines," Journal of the American Society of Naval Engineers, Vol. 58, pp. 493-508, Aug. 1946.
19. McMULLEN, JOHN J.: "A Free Piston Propulsion Plant For a Liberty Ship," ASME Paper #55-OGP-14, presented in Washington, D. C., June 1955.
20. UNDERWOOD, A. F.: "The GMR-4-4 'Hyprex' Engine — A Concept of the Free Piston Engine for Automotive Use," SAE Transactions, Vol. 65, p. 377, 1957.
21. NOREN, OSCAR B.: "A Free-Piston Tractor Power Plant." ERWIN, ROBERT L.: "Free-Piston Turbine Power For Farm Use," SAE Paper #150, presented at the SAE Summer Meeting in Atlantic City, N. J., June 2-7, 1957.
22. FREY, D. N., KLOTSCH, P., & EGLI, A.: "The Automotive Free Piston Engine," SAE Transactions, Vol. 65, p. 628, 1957.
23. OLSON, D. R.: "Simulation of a Free-Piston Engine With a Digital Computer," paper presented at SAE Meeting in Detroit, Jan. 13-17, 1958.
24. BOBROWSKY, A. R.: "Analytical Methods for Performance Estimates of Free-Piston Gasifiers," ASME Paper #57-GTP-6, presented at ASME Meeting in Detroit, Mich., March 18-21, 1957.
25. MORAIN, W. A. & SOO, S. L.: "Some Design Aspects of The Free Piston Gas Generator — Turbine Plant — Part I and Part II Controls and Accessories," ASME Papers #55-A-146 and 55-A-155 presented in Chicago, Ill., Nov. 13-18, 1955.
26. EICHELBERG, G.: "The Free Piston Gas Generator," Translated from Schweiz Baueitung, Nr. 48 and 49, Printed by Jean Frey AG, Zurich, 1948.
27. HUBER, R.: "Operational Stability of Free Piston Gasifiers," Journal of the American Society of Naval Engineers, Vol. 62, pp. 259-272, May 1950.
28. MOIROUX, A. F.: "The Free-Piston-Engine Possibilities," paper presented at ASME Oil & Gas Power Div. Meeting in Philadelphia, May 11-15, 1958.
29. BARTHALON, M. & HORGEN, H.: "French Experience With Free-Piston Gasifiers," ASME Paper #56-A-209, presented at ASME Meeting in New York, Nov. 25-30, 1956.
30. AUGUSTIN-NORMAND, JR., M. A. & BARTHALON, M.: "Some Problems in Design and Use of Free Piston Gas-Generators on Board Ships," paper presented at the Meeting of the Institution of Naval Architects in London, Dec. 6, 1956.
31. PICARD, F.: "An Experimental Turbo-Diesel Locomotive," Proc. 1, Mech. E., Vol. 169, No. 40, pp. 820-843, 1954.
32. SCANLAN, J. A. & JENNINGS, B. H.: "Bibliography of Free-Piston Engines and Compressors," ASME Paper #56-A-23 presented at ASME in New York, Nov. 25-30, 1956.
33. SAWYER, JOHN W.: "Marine Gas Turbine, Free Piston Gas Turbine Bibliography," Journal of the American Society of Naval Engineers, pp. 159-165, Feb. 1958.
34. MUNTZ, F. A. I. & BARTHALON, M.: "Free-Piston Progress," The Motor Ship, pp. 492-494, Jan. 1958.
35. BRADLEY, W. F.: "French Gas Turbine Truck Has Two Free-Piston Engines," Automotive Industries, Vol. 106, pp. 52, 106, Feb. 15, 1952.
36. GILLINGHAM, A. F.: "Free Piston Engines for the Oil Industry," The Petroleum Times, p. 763, Aug. 30, 1957.
37. PESCARA, R. P.: "Free-Piston Machines," Journal de la SIA, Vol. 10, pp. 423-433, 1937 (in French).
38. Anon, Booklet on "Free Piston Diesel Engines," published by "SIGMA," 61, Avenue Franklin-D-Roosevelt, Paris, France, 1955.
39. Anon, "Free Piston Boosts Air Conditioning," American Gas Association Monthly, June 1958, Page 16.

Printed in U. S. A. by
Salley & Collins, Inc.
305 East 45th Street
New York 17, N. Y.